

Investigation of Mixture Formation and Combustion in an Ethanol Direct Injection plus Gasoline Port Injection (EDI+GPI) Engine

By

Yuhan Huang

A thesis in fulfilment of the requirements for the degree of
Doctor of Philosophy

School of Electrical, Mechanical and Mechatronic Systems
Faculty of Engineering and Information Technology
University of Technology Sydney

December 2016

Certificate of Original Authorship

This thesis is the result of a research candidature conducted jointly with another university as part of a collaborative doctoral degree. I certify that the work in this thesis has not previously been submitted for a degree nor has it been submitted as part of requirements for a degree except as part of the collaborative doctoral degree and/or fully acknowledged within the text.

I also certify that the thesis has been written by me. Any help that I have received in my research work and the preparation of the thesis itself has been acknowledged. In addition, I certify that all information sources and literature used are indicated in the thesis.

Signature of Student:

Date:

Acknowledgements

To pursue a doctoral degree could be a long and challenging journey. Through this journey, I fortunately received help and support from the following wonderful people who made this journey enjoyable and fruitful.

First of all, I would like to thank my principle supervisor Associate Professor Guang Hong who provided huge support and guidance. She invested numerous efforts in supervising me and always cared about my progress and future career. The experience I have acquired and research training I have received from her will greatly benefit my research career. I would also like to thank my co-supervisor Professor Ronghua Huang who supported the spray experiments at the Huazhong University of Science and Technology (HUST). Furthermore, I appreciate the discussion and support received from Dr. Jack Wang, Professor Xiaobei Cheng and Dr. Zhaowen Wang.

I really appreciate Nizar Al-Muhsen and Yuan Zhuang for your time and assistance in the engine experiments. The ethanol fuel provided by the Manildra Group is greatly acknowledged. Many thanks to Peng Deng, Sheng Huang and Yinjie Ma for your suggestions and help in the spray experiments at HUST. I also appreciate Laurence Stonard, John Funnell, Peter Tawadros and Jack Liang for your excellent support in the maintenance of the engine. Thanks to Vahik Avakian and Matthew Gaston for your valuable support in the engine CFD modelling.

I gratefully thank my close friends Mahdi Hassan, Nizar Al-Muhsen, Sayed Royel, etc. for the wonderful time we spent together throughout my candidature at UTS. I wish you all health and success.

This study was sponsored by the dual doctoral degree program between UTS and HUST. The funding provided by the China Scholarship Council is gratefully appreciated.

Last but most importantly, I would like to thank my wife Jennifer Liu for her support and care. No matter how tired I was in the office, I always felt warm and relaxed once I got home. With you by my side, there is no difficulty I could not cope with and there is no dream I could not realise. Meanwhile, I would also like to thank my parents and parents-in-law for their love and support.

List of Publications

Journal articles

- [1] **Y. Huang**, G. Hong. Investigation of the effect of heated ethanol fuel on combustion and emissions of an ethanol direct injection plus gasoline port injection (EDI + GPI) engine. *Energy Conversion and Management* 2016; 123: 338-347.
- [2] **Y. Huang**, G. Hong, R. Huang. Effect of injection timing on mixture formation and combustion in an ethanol direct injection plus gasoline port injection (EDI+GPI) engine. *Energy* 2016; 111: 92-103.
- [3] **Y. Huang**, S. Huang, R. Huang, G. Hong. Spray and evaporation characteristics of ethanol and gasoline direct injection in non-evaporating, transition and flash-boiling conditions. *Energy Conversion and Management* 2016; 108: 68-77.
- [4] **Y. Huang**, G. Hong, R. Huang. Investigation to charge cooling effect and combustion characteristics of ethanol direct injection in a gasoline port injection engine. *Applied Energy* 2015; 160: 244-254.
- [5] **Y. Huang**, G. Hong, R. Huang. Numerical investigation to the dual-fuel spray combustion process in an ethanol direct injection plus gasoline port injection (EDI+GPI) engine. *Energy Conversion and Management* 2015; 92: 275-286.
- [6] **Y. Huang**, S. Huang, P. Deng, R. Huang, G. Hong. The Effect of Fuel Temperature on the Ethanol Direct Injection Spray Characteristics of a Multi-hole Injector. *SAE Int. J. Fuels Lubr.* 2014; 7: 792-802.

Conference proceedings

- [7] **Y. Huang**, G. Hong. An Investigation of the Performance of a Gasoline Spark Ignition Engine Fuelled with Hot Ethanol Direct Injection. *Australian Combustion Symposium*, the Combustion Institute, Melbourne Australia; 2015.
- [8] **Y. Huang**, G. Hong, R. Huang. The Effect of Volume Ratio of Ethanol Directly Injected in a Gasoline Port Injection Spark Ignition Engine. *10th Asia-Pacific Conference on Combustion*, the Combustion Institute, Beijing China; 2015.

- [9] **Y. Huang**, S. Huang, R. Huang, G. Hong. Macroscopic and Microscopic Characteristics of Ethanol and Gasoline Sprays. *19th Australasian Fluid Mechanics Conference*, Melbourne Australia; 2014.
- [10] **Y. Huang**, G. Hong. Development of a Numerical Model for Investigating the EDI+GPI Engine. *19th Australasian Fluid Mechanics Conference*, Melbourne Australia; 2014.
- [11] **Y. Huang**, G. Hong, R. Huang. Numerical Investigation to the Effect of Ethanol/Gasoline Ratio on Charge Cooling in an EDI+GPI Engine. *SAE paper 2014-01-2612*; 2014.
- [12] **Y. Huang**, G. Hong, X. Cheng, R. Huang. Investigation to Charge Cooling Effect of Evaporation of Ethanol Fuel Directly Injected in a Gasoline Port Injection Engine. *SAE paper 2013-01-2610*; 2013.

Abstract

Ethanol direct injection plus gasoline port injection (EDI+GPI) is a new technology to utilise ethanol fuel in spark-ignition engines more effectively and efficiently than E10 or E85 fuels in the current market. It takes the advantages of ethanol's high octane number and great enthalpy of vaporisation which allow higher compression ratio and consequently increase the thermal efficiency. Primary experimental investigation showed that the engine performance was improved by EDI+GPI. The thermal efficiency was increased, the NO emission was decreased and the spark timing could be advanced without engine knock. However, the CO and HC emissions were increased when EDI was applied. To understand the mechanisms behind the experimental results, the mixture formation and combustion processes of an EDI+GPI engine were investigated using CFD simulation, and constant volume chamber and engine experiments.

To investigate the spray and evaporation characteristics of ethanol fuel and provide experimental data for CFD simulation, spray experiments were conducted in a constant volume chamber using high speed shadowgraphy imaging technique. The results showed that ethanol fuel evaporated slowly when fuel temperature was in the range of 275-325 K. However, the evaporation rate increased quickly when fuel temperature was higher than 350 K. The low evaporation rate of ethanol fuel in low temperature environment implied that EDI should be only applied in high temperature engine environment. When the excess temperature was smaller than 4 K, the spray behaved the same as the subcooled spray did. The spray collapsed when the excess temperature was 9 K. Flash-boiling did not occur until the excess temperature reached 14 K.

Numerical simulation of the EDI+GPI engine showed that the overall cooling effect of EDI was enhanced with the increase of ethanol ratio from 0% to 58%, but not with further increase of ethanol ratio. When the ethanol ratio was greater than 58%, the fuel impingement became severe and a large number of liquid ethanol droplets were left in the combustion chamber during combustion, leading to local over-cooling in the near-wall region and over-lean mixture at the spark plug gap. As a consequence, the CO and HC emissions increased due to incomplete combustion. Compared with GPI only condition, the faster flame speed of ethanol fuel in EDI+GPI condition resulted in shorter combustion initiation duration and major combustion duration, leading to the increase of IMEP and thermal efficiency when the ethanol ratio was 0-58%. However, the

combustion performance was deteriorated by over-cooling and fuel impingement when ethanol ratio was greater than 58%. Experimental results showed consistently that the combustion and emission performance of this engine could be the best in the ethanol ratio of 40-60% at the investigated engine condition (medium load, 4000 rpm and early EDI timing of 300 CAD BTDC). Numerical results showed that the best engine performance was resulted from effective charge cooling and combustion efficiency improved by avoiding the wall wetting, over-lean and local over-cooling issues. Numerical simulations were also carried out to investigate the effect of direct injection timing on the EDI+GPI. The results showed that when the EDI timing was retarded from 300 to 100 CAD BTDC, the mixture around the spark plug became leaner and the distribution of equivalence ratio became more uneven. Moreover, late EDI timing at 100 CAD BTDC resulted in severe fuel impingement and caused local over-cooling effect and over-rich mixture. Consequently, the combustion speed and temperature were decreased by retarded EDI timing, leading to the decreased NO emission and the increased HC and CO emissions. The fuel impingement and incomplete combustion of late EDI timing at 100 CAD BTDC could be addressed by reducing the ethanol ratio to an appropriate point.

Experiments on the EDI+GPI engine were conducted to verify the idea of EDI heating on improving the engine performance, which was developed based on the understanding gained from the numerical investigation. Results showed that EDI heating effectively reduced the CO and HC emissions at the original engine's spark timing of 15 CAD BTDC. Meanwhile, the NO emission was slightly increased, but still much smaller than that in GPI only condition. However, the IMEP and combustion speed were slightly reduced by EDI heating. To enhance the effect of EDI heating, experiments were conducted at varied spark timing. The results at the MBT timing (19 CAD BTDC) showed that the reduction of IMEP by EDI heating was less significant whilst the CO and HC emissions were effectively reduced. Therefore EDI heating was effective to address ethanol's low evaporation rate and over-cooling effect issues in the development of EDI+GPI engine in terms of minimizing the emissions.

Contents

Certificate of Original Authorship.....	i
Acknowledgements	ii
List of Publications.....	iii
Abstract.....	v
Contents	vii
List of Tables	x
List of Figures.....	xi
Definitions and Abbreviations.....	xvi
1 Introduction.....	1
1.1 Research background and motivation	1
1.2 Research methodology and objectives	2
1.3 Thesis outline	3
2 Literature Review	5
2.1 Development of EDI+GPI.....	5
2.2 CFD engine modelling	8
2.2.1 Turbulence modelling	9
2.2.2 Spray modelling	10
2.2.3 Combustion modelling.....	17
2.3 Fuel properties and effect on spray combustion.....	26
2.3.1 Fuel properties.....	26
2.3.2 Spray characteristics of ethanol and gasoline fuels.....	30
2.3.3 Measuring the charge cooling effect	34
2.3.4 Combustion characteristics of ethanol fuel	36
2.3.5 Fuel heating to generate fine and fast evaporating sprays	40
3 Experimental Setup and CFD Model	43
3.1 Test rig for investigating EDI sprays	43
3.1.1 EDI injector.....	43
3.1.2 Constant volume chamber optical system.....	44
3.1.3 Test fuel.....	46

3.1.4	Experimental conditions.....	47
3.1.5	Image Processing	47
3.2	CFD model of EDI+GPI engine.....	50
3.2.1	Computational mesh	51
3.2.2	Spray sub-models	53
3.2.3	Combustion sub-models.....	55
3.2.4	Boundary and initial conditions	58
3.2.5	Model verification.....	59
3.3	Test rig for investigating EDI heating.....	64
3.3.1	Engine setup and ethanol heating.....	64
3.3.2	Instruments and measurements	66
3.3.3	Test fuels	67
3.3.4	Engine experiment conditions.....	67
3.3.5	In-cylinder pressure data processing.....	69
4	EDI Spray Characteristics.....	71
4.1	Characteristics of non-evaporating and evaporating sprays.....	71
4.2	Characteristics of flash-boiling sprays	75
4.3	Spray transition process.....	78
4.4	Implications for engine emissions.....	80
4.5	Summary	81
5	Mixture Formation and Cooling Effect of EDI+GPI.....	83
5.1	EDI+GPI sprays	83
5.2	Cooling effect of EDI.....	87
5.3	Effect of ethanol ratio on charge cooling	93
5.4	Summary	100
6	Combustion and Emission characteristics of EDI+GPI	102
6.1	Combustion and emissions of EDI+GPI	102
6.2	Effect of ethanol ratio on combustion	109
6.3	Effect of EDI timing on fuel evaporation, mixing and combustion	115
6.4	Effect of small ethanol ratio on reducing fuel impingement.....	123
6.5	Summary	125

7	Effect of EDI Heating on Engine Performance	128
7.1	Engine performance of EDI heating at original spark timing	128
7.2	Combustion characteristics of EDI heating at original spark timing	133
7.3	Engine performance of EDI heating at MBT timing.....	138
7.4	Summary	142
8	Conclusions and Future Work	144
8.1	Conclusions	144
8.2	Suggestions for future work	147
	References	149
	Appendix	164

List of Tables

Table 2.1: The cost of various simulation approaches [57]	10
Table 2.2: Physical and chemical properties of ethanol and gasoline fuels.....	26
Table 2.3: Experimental conditions in [45] and [117]	39
Table 3.1: Specifications of the EDI+GPI engine.....	65
Table 3.2: EDI heating experiment conditions	68
Table A.1: List of engine simulation conditions and related publications.....	164
Table A.2: List of spray experiment conditions and related publications.....	165
Table A.3: List of engine experiment conditions and related publications.....	166

List of Figures

Figure 2.1: The evolution processes of a liquid fuel spray [60].....	12
Figure 2.2: Different types of secondary breakup [75].....	14
Figure 2.3: Droplet distortion.....	15
Figure 2.4: Schematics of laminar (left) and turbulent (right) Bunsen premixed flames [85].....	18
Figure 2.5: Combustion regimes of turbulent premixed flames [86].....	20
Figure 2.6: Schematics of laminar (a) and turbulent (b) non-premixed flames[55]	21
Figure 2.7: Combustion regimes of turbulent non-premixed flames, (a) [55], (b) [85]..	21
Figure 2.8: Schematics of combustion styles with different overall equivalence ratios (ϕL is the lean flammability limit and ϕR is the rich flammability limit) [87]	22
Figure 2.9: Parameters for describing the premixed and non-premixed flames [55].....	24
Figure 2.10: Saturation vapour pressures of gasoline, ethanol and iso-octane fuels	27
Figure 2.11: Comparison between the ethanol vapour pressures calculated from the experimental data [101] and Yaws' handbook [100]	28
Figure 2.12: Laminar flame speeds of ethanol and gasoline fuels [102]	29
Figure 2.13: Primary breakup structures of gasoline (a) and ethanol (b) sprays [111]...	32
Figure 3.1: Overall view of the EDI injector tip (left) and the magnified image of the nozzle holes (right).....	44
Figure 3.2: Lateral view of the spray plumes and their directions.....	44
Figure 3.3: Front view of the spray plumes and their footprints.....	44
Figure 3.4: Schematic of the EDI spray experiment apparatus.....	45
Figure 3.5: Ethanol fuel properties (data source: [100]).....	46
Figure 3.6: Shadowgraphy spray image processing method.....	48
Figure 3.7: Effect of the threshold vaule on the detected spray boundary.....	49
Figure 3.8: Cross-sectional view of the geometry of the cylinder head.....	51

Figure 3.9: Geometry decomposition.....	52
Figure 3.10: Computational mesh	52
Figure 3.11: Measured valve lifts of the intake and exhaust valves	52
Figure 3.12: Computational meshes with different grid densities	60
Figure 3.13: Grid size sensitivity	61
Figure 3.14: Comparison of the experimental and numerical results of EDI spray pattern at 1.5 ms ASOI in a constant volume chamber @ 6.0 MPa injection pressure, 0.1 MPa ambient pressure and 350 K ambient temperature	61
Figure 3.15: Comparison of the experimental and numerical results of EDI spray tip penetration in a constant volume chamber @ 6.0 MPa injection pressure, 0.1 MPa ambient pressure and 350 K ambient temperature	62
Figure 3.16: Comparison between the measured and simulated values of in-cylinder pressure and heat release rate at different ethanol ratios.....	63
Figure 3.17: Schematic of the EDI+GPI engine	64
Figure 3.18: Schematic of EDI fuel heating system	65
Figure 3.19: logarithmic pressure-volume diagram [184]	70
Figure 4.1: Spray images in non-evaporating and evaporating conditions	72
Figure 4.2: Averaged pixel intensity of the spray area at 8 ms ASOI for sprays in non-evaporating and evaporating conditions.....	73
Figure 4.3: Spray tip penetrations in non-evaporating and evaporating conditions	73
Figure 4.4: Spray projected areas in non-evaporating and evaporating conditions	74
Figure 4.5: Spray angles in non-evaporating and evaporating conditions	74
Figure 4.6: Spray images in flash-boiling conditions (Please refer to the electronic version of this figure for a clearer interpretation of the droplet explosion).....	76
Figure 4.7: Schematic of the droplet flash-boiling process	76
Figure 4.8: Averaged pixel intensity of the spray area at 8.0 ms ASOI for sprays in flash-boiling conditions.....	77
Figure 4.9: Spray tip penetrations in flash-boiling conditions	77

Figure 4.10: Spray projected areas in flash-boiling conditions.....	78
Figure 4.11: Spray angles in flash-boiling conditions	78
Figure 4.12: Spray images in the transition process (Please refer to the electronic version of this figure for a clearer interpretation of the spray cloud)	79
Figure 5.1: Spatial distributions of spray droplets and air flow velocity vectors on the engine symmetry plane of EDI+GPI E46 at: (a) the intake TDC, (b) the end of GPI injection, (c) 15 CAD after the start of EDI, and (d) at the spark timing	84
Figure 5.2: Structure of the EDI spray at: (a) 15 CAD after the start of EDI injection, (b) the end of EDI injection and (c) the BDC.....	85
Figure 5.3: Distributions of gasoline and ethanol fuel mass fractions and overall equivalence ratio by spark timing	86
Figure 5.4: Cooling effect enhancement of EDI: (a) comparison of in-cylinder mean temperature in E0, E46 and E100 conditions, (b) the temperature reduced by EDI.....	88
Figure 5.5: Mass of the injected and evaporated fuels (a) and the enthalpy for their evaporation (b) in E0, E46 and E100 conditions	90
Figure 5.6: Saturation vapour pressures of gasoline [101] and ethanol [100] fuels	91
Figure 5.7: In-cylinder temperature distribution on a vertical plane passing through the spark plug predicted for GPI only E0 (a), EDI+GPI E46 (b) and EDI only E100 (c) conditions at the spark timing	92
Figure 5.8: In-cylinder temperature distribution on a horizontal plane predicted for GPI only E0 (a), EDI+GPI E46 (b) and EDI only E100 (c) conditions at the spark timing ..	93
Figure 5.9: In-cylinder temperature distributions by spark timing	94
Figure 5.10: Ethanol droplet distributions of EDI+GPI E46 at the end of EDI (left) and spark timing (right)	95
Figure 5.11: Variation of mean in-cylinder temperature by spark timing with the ethanol ratio	95
Figure 5.12: Completeness of the ethanol and gasoline evaporation by spark timing....	96
Figure 5.13: Ideal cooling potential and charge cooling realized	97
Figure 5.14: Distribution of equivalence ratio around the spark plug by spark timing ..	98

Figure 5.15: Distributions of the ethanol spray droplets at the end of EDI injection	99
Figure 6.1: Flame propagation in GPI only E0 and EDI+GPI E46	103
Figure 6.2: In-cylinder temperature in GPI only E0 and EDI+GPI E46 conditions	104
Figure 6.3: The mass of unburnt vapour (a) and liquid (b) fuels	106
Figure 6.4: Formation of NO (a) and CO (b) emissions during the combustion	107
Figure 6.5: Spatial distributions of temperature, O, OH, and NO at 405 CAD	109
Figure 6.6: Distributions of O ₂ and CO at 405 CAD	109
Figure 6.7: In-cylinder pressure varying with the ethanol ratio	110
Figure 6.8: IMEP varying with the ethanol ratio	110
Figure 6.9: CA0-10% and CA10-90% varying with the ethanol ratio.....	111
Figure 6.10: Flame propagation and distributions of OH mass fraction at 375 CAD and 395 CAD varying with the ethanol ratio	112
Figure 6.11: The distributions of ethanol liquid droplets, equivalence ratio and cylinder temperature at 395 CAD varying with the ethanol ratio	113
Figure 6.12: Measured engine emissions varying with the ethanol ratio.....	114
Figure 6.13: Experimental results of IMEP, ISNO, ISCO and ISHC varying with EDI timing	115
Figure 6.14: Variations of mass of the vapour ethanol fuel with crank angle degrees at three injection timings.....	116
Figure 6.15: Air flow velocity vectors on the engine symmetry plane at the start and end of EDI injection with different EDI timings	117
Figure 6.16: Distributions of the vapour mass fractions of gasoline and ethanol fuels and the equivalence ratio on a vertical plane passing through the spark plug by spark timing	118
Figure 6.17: Variation of wall film mass with crank angle degrees	119
Figure 6.18: Evolution of flame-brush with different EDI timings	119
Figure 6.19: Spatial distributions of combustion temperature with different EDI timings	120

Figure 6.20: Distributions of flame brush, ethanol droplets, equivalence ratio and combustion temperature at EVO with different EDI timings	121
Figure 6.21: Distribution of NO mass fraction at EVO	122
Figure 6.22: Distribution of CO mass fraction at EVO	123
Figure 6.23: Distributions of the equivalence ratio and wall film of smaller amount of ethanol fuel at IT100 by spark timing	124
Figure 6.24: Distributions of flame brush, ethanol droplets and in-cylinder temperature of smaller amount of ethanol fuel at IT100 at the time of EVO.....	125
Figure 7.1: Effect of ethanol fuel temperature on the IMEP.....	129
Figure 7.2: Effect of ethanol fuel temperature on the indicated thermal efficiency	130
Figure 7.3: Variation of ISCO with ethanol fuel temperature	131
Figure 7.4: Variation of ISHC with ethanol fuel temperature	131
Figure 7.5: Variation of ISNO with ethanol fuel temperature	133
Figure 7.6: Effect of ethanol fuel temperature on the in-cylinder pressure and heat release rate.....	133
Figure 7.7: Variation of lambda with ethanol fuel temperature.....	135
Figure 7.8: Effect fuel temperature on the EDI spray structure.....	136
Figure 7.9: Variation of CA0-10% with ethanol fuel temperature	137
Figure 7.10: Variation of CA10-90% with ethanol fuel temperature	138
Figure 7.11: Effect of spark timing on IMEP (a) and NO emission (b) at ethanol ratios of 0%, 55% and 100%.....	139
Figure 7.12: Effect of EDI heating on IMEP at the MBT timing	140
Figure 7.13: Effect of EDI heating on ISCO at the MBT timing.....	141
Figure 7.14: Effect of EDI heating on ISHC at the MBT timing.....	141
Figure 7.15: Effect of EDI heating on ISNO at the MBT timing	142

Definitions and Abbreviations

Acronyms

ABDC	After bottom dead center
ASOI	After the start of injection
ATDC	After top dead center
BBDC	Before bottom dead center
BDC	Bottom dead center
BTDC	Before top dead center
CAD	Crank angle degrees
CFD	Computational fluid dynamics
DI	Direct injection
ECFM	Extended Coherent Flame Model
EDI	Ethanol direct injection
EDI+GPI	Ethanol direct injection plus gasoline port injection
EVC	Exhaust valve close
EVO	Exhaust valve open
GDI	Gasoline direct injection
GPI	Gasoline port injection
IC	Internal combustion
IMEP	Indicated mean effective pressure
ITNFS	Intermediate turbulent net flame stretch
IVC	Intake valve close
IVO	Intake valve open
MBT	Minimum spark advance for best torque
MFB	Mass fraction burnt
PDF	Probability Density Function
PI	Port injection
RANS	Reynolds Averaged Navier-Stokes
SI	Spark ignition
TDC	Top dead center

Symbols

A_p	Particle surface area
C_D	Drag coefficient
D_i	Diffusion coefficient in air
N_i	Molar flux of vapour
P	Pressure
D	Dissipation term of flame area
D_i	Diffusion coefficient in air
P_1	Source term due to turbulence interaction
P_2	Source term due to dilatation in the flame
P_3	Source term due to expansion of burned gas
P_4	Source term due to normal propagation
T	Temperature
U_L	Laminar flame speed
V	Volume
X_i	Mole fraction of species i
Y_i	Mass fraction of species i
Z	Mixture fraction
B_m	Spalding mass number
Da	Damköhler number
Ka	Karlovitz number
Re	Reynolds number
We	Weber number
ϕ	Fuel/air equivalence ratio
Σ	Flame area density
Γ_K	ITNFS term
c	Progress variable
c_p	Heat capacity
d	Diameter
h	Heat transfer coefficient
m	Mass

k	Turbulent kinetic energy
r	Radius
y	Distortion of the droplet
γ	Specific heat ratio
ε	Turbulent dissipation rate
σ	Surface tension
ρ	Density
μ	Dynamic viscosity
ν	Kinematic viscosity
t	Time
μ_t	Turbulent viscosity
u	Velocity
u'	Turbulent velocity fluctuation
l_t	Integral turbulent length scale
l_d	Diffusion thickness
l_r	Reaction zone thickness
τ_t	Turbulent time scale
τ_c	Chemical time scale
τ_k	Kolmogorov time scale
k_c	Mass transfer coefficient
δ_l	Flame thickness
$\varphi_{realized}$	Percentage of charge cooling realized
$CA0-10\%$	Combustion initiation duration
$CA10-90\%$	Major combustion duration
Pa/Ps	Ambient-to-saturation pressure ratio
$E'X'$	X% ethanol by volume. e.g. E46 is 46% ethanol via direct injection plus 54% gasoline via port injection
$IT'XXX'$	Injection timing of XXX CAD BTDC
ΔT	Spray excess temperature
ΔT_{actual}	Actual cooling effect
ΔT_{ideal}	Ideal cooling potential

Subscripts

d	Droplet phase
g	Gas phase
i	Species i
l	Liquid phase
p	Particle
rel	Relative
sat	Saturation
∞	Ambient bulk gas